

FINITE ELEMENT ANALYSIS OF GAS FOIL BEARINGS

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CERTIFICATE

This is to certify that the thesis entitled, “**Finite Element Analysis of Gas Foil Bearings**” submitted by Mr. Preetiranjana Pradhan (111ME0335) in partial fulfillment of the requirements for the award of Bachelor of Technology in Mechanical Engineering at the National Institute of Technology, Rourkela is an authentic work carried out by them under my supervision and guidance. To the best of my knowledge, the matter embodied in the thesis has not been submitted to any other University/ Institute for the award of any Degree or Diploma.

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ABSTRACT

Micro turbomachinery demands gas bearings to be light compact and should operate at varying temperature conditions. Low heat generation friction and lack of lubricant circulation system makes it compact reliable and eco-friendly. However low stiffness and damping coefficients, high cost and lack of sufficient knowledge and predictive tools somewhat restricts GFBs use in mass produced application. Current commercial and engineering application demands more and more aggressive designs with high surface speed and unit loads as well as thinner fluid film. Again rotor-dynamics analysis uses stiffness and damping coefficients to represent fluid film behavior or in other words these coefficients play the key role in determining dynamic characteristics of a rotor shaft. Stiffness coefficients depend mainly on two factors first the static deflection of foil due to shaft load and second the hydrodynamic effect produced due to the fluid film. Here is an approach to calculate the stiffness coefficient produced due to static deflection of GFBs due to static load using finite element analysis and the stiffness coefficient has been calculated. Reynold's equation is to be solved using FDM to obtain pressure profile during hydrodynamic action of fluid film and using these pressure values in the bearing model dynamic component of stiffness can be produced. Adding both components will produce the overall stiffness coefficient of a gas foil bearing.

Key words: FEA, gas foil bearing, stiffness coefficients.

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CHAPTER1: INTRODUCTION

1.1 ROTOR DYNAMICS

It's the branch of engineering that focuses on vibrations of rotating shafts with objective being predicting the vibration and keeping it below the acceptance level. The principal components are a shaft or a rotor with a disk and a bearing. The shaft is the rotating component of the system and many industrial applications have shafts designed in a long and thin geometry to provide optimum space for components such as seals and impellers. The bearing supports the rotating shaft of the system and provide damping to stabilize the system and limit vibration to certain level.

1.2 BEARINGS

A bearing is a mechanical device that separates two opposing surfaces completely by a layer of fluid lubricant. Plain journal bearing is used commercially in almost all devices that has a rotating part. Compressors, pumps turbines, motors generators are few examples need to be mentioned. Journal bearing is a structure where two cylinders rotate concentrically relative to each other. One being the shaft, rotating at a particular angular speed, and other the bearing. The main objective is to support the rotating structures and provide sufficient lubrication to avoid friction that causes wear and tear to machine parts. The fluid film at high pressure provides the hydrodynamic film lubrication and determines the load capacity of the bearing. Eccentricity is a bearing parameter which is defined as the displacement between shaft and bearing center.

1.3 GAS FOIL BEARINGS

Foil bearings or gas foil bearings (GFB) comes under the category of air bearings. Here the shaft is supported by a thin foil journal lining. Once the shaft moves at high speed the fluid film pushes the shaft away to avoid contact. Shaft and foil are separated by high pressure fluid film (here ambient air in most cases) generated by rotation which pulls air into the bearing via viscosity effects. High speed of rotor with respect to the bearing creates this initial air gap which prevents wear and tear. Unlike aero and hydrostatic bearings GFBs do not need external pressurized system for working fluid hence these are self-starting bearings.

Micro turbomachinery demands gas bearings to be light compact and should operate at varying temperature conditions. Gas bearings offer reduced heat generation low friction and don't require sealing and lubricant circulation system hence are compact, reliable and light weight also they eliminate process fluid contamination [1] and are sustainable and ecofriendly, therefore GFBs with high stiffness and damping coefficients and low cost will enable it for commercial usage; but high cost and lack of sufficient knowledge and predictive tools have denied use of GFBs in mass produced applications. Low load carrying capability and minute film thickness causes its fabrication and installation costs money as well as time. Poor damping capacity due to low viscosity of air is another disadvantage.

TYPES OF GAS FOIL BEARINGS:

Bearings in rotating machinery can be

- a. Radial bearings supporting lateral loads including rotor weight
- b. Thrust bearing carrying axial loads

Few other configurations are shown here:

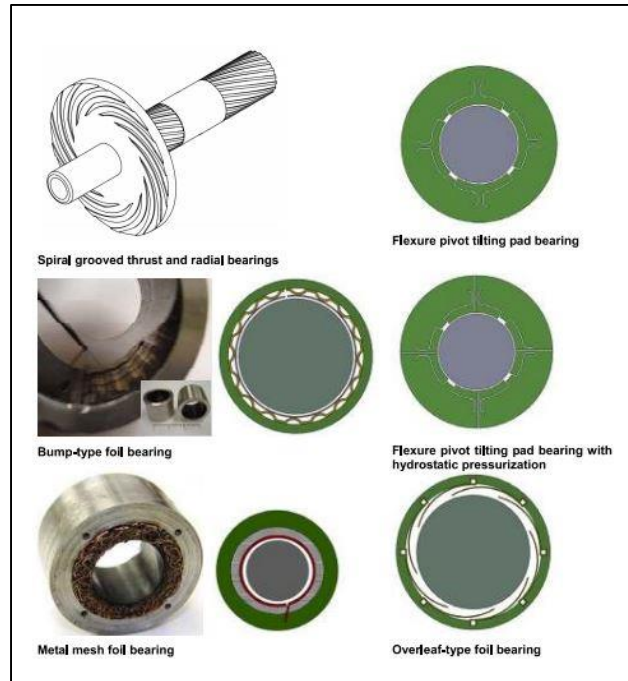


Figure 1: Typical commercial gas foil bearings for micro-turbomachinery [2].

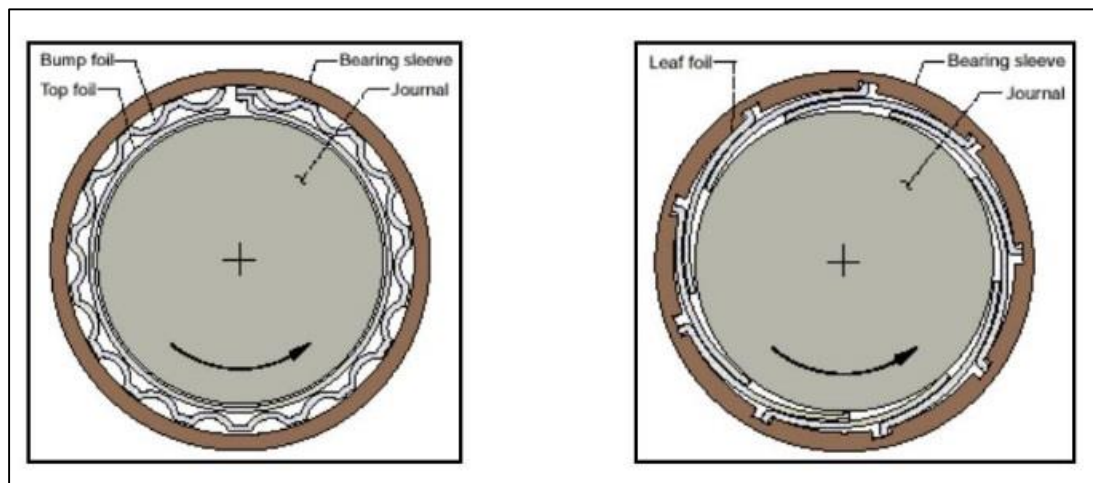


Figure 2: Commercially used bump foil and leaf foil bearings
(source://<http://www.grc.nasa.gov/WWW/oilfree/bearings.htm>)

In bump-type GFBs the top foil is supported by series of compliant bumps and deforms elastically under hydrodynamic fluid film pressure. The bearing stiffness is a combined result from the bump deflection and also by the hydrodynamic film generated during the shaft rotation. Coulomb type damping arises between the bumps and the top foil or between the bumps and bearing wall. GFBs also allow larger misalignments of shafts as the foils comply and deflect under misalignment. All sorts of misalignments caused due to tolerance build-ups, thermal expansion and centrifugal shaft growth are absorbed at the cost of expansion of the top foil. GFBs generally uses ambient air but lubricants like LOX and inert gases can also be used for specific application

Currently they find their applications in air-cycle units in aircrafts, cryogenic turbo expanders for gas separation plants, high precision instruments, power units of aerospace as well as ground vehicles, gas turbines of automobiles, compressors and MTM.

1.4 FINITE ELEMENT ANALYSIS

FEA is a numerical technique developed to solve complex problems dealing with varying shape, boundary conditions and loads. The basic idea is to replace the complicated problem by a simpler one hence the solutions obtained are approximate. The existing tools might not be sufficient to obtain the exact solution or even an approximate solution. This leads us to FEA to solve such problems, however more computational effort can be provided to refine the approximate solution. This originated as a method of stress analysis in designing aircrafts but today we can see it application a varying fields such as solid mechanics, fluid flow heat transfer, design of civil structures, air crafts, space crafts, electric motors and heat engines. Finite element analysis is very much versatile and can handle both static and dynamic problems.

In engineering we have to find few parameters or field variables, once these are obtained behavior of entire structure can be calculated. These field variables may be displacements in solid mechanics and velocities in fluid. In a continuum these are infinite. FEA reduces these field variables to a finite number by dividing solution region into smaller elements and field variables are assumed as approximate functions within each elements. These functions are defined in terms of the unknowns or field variables of specified points called nodal points or nodes. Thus in FEA the unknowns are the field variables of nodal points, once they are obtained the field variable at any point can be calculated using interpolation functions.

ADVANTAGES OF FEA:

- i. Solutions for all sorts of problem can be obtained although they are approximate.
- ii. It can handle materials with anisotropic properties without any difficulty.
- iii. Solutions of problems involving critical geometry and critical boundary conditions can be solved easily.
- iv. If structure comprises more than one material then FEA is more reliable than any other method.
- v. Problems with material and geometric non-linearity can be handled with ease by FEA.

CHAPTER 2: LITERATURE SURVEY

2.1 LITERATURE REVIEW

Carpino and peng [3] calculated stiffness and damping coefficients of a foil bearing supported elastically. Ambient gas is used as fluid film and properties are obtained by Reynold's equations. Structural model is designed and both the equations fluid and structural are coupled and perturbation method is deployed to obtain dynamic equations. FDM has been used to calculate all coefficients and it is found that the elastic foundation reduces these coefficients and these coefficients are very much important in determining rotor shaft stability and calculations of rotor dynamics. As this methods calculates all above coefficients in general manner from perturbation equations numerical errors are less and accuracy is more.

Heshmat et al. [4] used a spring supported compliance bearing to evaluate performance of a GFB. Analysis of effect of various structural and geometric variables on bearing performance and behavior was done. Solution of relevant differential equations includes values of spring coefficients due to both structural and hydrodynamic stiffness. Desirable design features are discussed. A comparative study of stability of GFBs shows that bearings can be designed to a desired stiffness to fulfill demands of its rotor dynamics system.

Heshmat et al. [5] later observed that load performance of GFB is related to bearing compliance and hydrodynamics of wedge surface, where compliance comprises elastically foundation and top foil. They combined finite difference and finite element method to investigate a class of gas foil thrust bearing analytically. Hydrodynamic equation is solved and solution is coupled with structural resiliency. And FD and FE method are used to solve above mentioned problem respectively. Elasticity effects of this combined system is addressed by the influence coefficients

and convergence was obtained. Application of FDM and FEA coherently proved to be a great methodology to handle analysis of GFBs.

Czolczyński [6] for analyzing motion of machines like GFBs believed it is convenient to represent dynamic property by means of stiffness and dynamic coefficients. He represented GFB by a mathematical model based on Reynold's equations and other few equations to describe gas flow in feeding systems. The coefficients obtained are functions of bearing parameters, external load, angular velocity of shaft and its frequency of vibration they also determine stability resonance vibration and amplitude of unbalanced vibration. He also pointed out that during solving the equations the larger distance between grid points (for calculating coefficients) lead to significant between real values and values of the curve fitted polynomial.

Lee et al. [7] investigated dynamic performance of GFBs at high temperatures experimentally. A GFB was mounted on a rotating shaft, which was hollow, a cartridge heater heated the shaft from inside and the shaft was excited by two electromagnetic shakers orthogonally to determine the frequency dependent stiffness and damping coefficient in a dynamic performance test rig. A set of temperatures and frequency were taken for conducting the experiment whereas load and rpm kept constant. They identified that cross coupled coefficients of a GFB are smaller than direct coefficients.

Park et al. [8] investigated rotor-dynamics of a micro power system supported by GFBs. FEM was used to perform stability analysis with predicted dynamic coefficients of the foil bearing. A test rig was prepared to simulate operating characteristics for e.g. Speed up to 300k rpm was considered and results for unbalance response were compared. They finally showed stable performance can be achieved using a GFB.

Miaskowski [9] presented selected numerical models being used for modelling gas foil bearings all around the universe. Difficulties arising due to complex structure of GFB are addressed. For analyzing a GFB the structural base hydrodynamic effect and their interaction is to be considered. Current modelling are time consuming and costly hence a reliable method is required to contribute significantly to bearing design.

Jamir et al. [10] noticed the poor load carrying tendency of gas foil bearings and provided a numerical method to obtain the performance of GFBs. FDM is employed to get non-dimensional form and discretize Reynold's equation and pressure profile is generated using numerical methods (here Newton-Raphson technique). Static performance test are carried out and compared with results presented earlier. Their study pointed out the effect of low load carrying capacity due to thinner fluid film and need of detail foil structure for more finite element analysis.

Andres and Wilde [11] advanced for a novel finite element method for analyzing gas foil bearings. To prevent non symmetrical weigh function they introduced a higher order weigh function that naturally incorporates flow character. This FE formulation model proves to be accurate fast, reliable and efficient however neglects impact due to high speed operations.

Andres and Kim [12] noticed that performance of GFB depends largely on its top foil and bump foil which is the elastic structure of a foil bearing. Age old methods include these foils as an equivalent stiffness uniformly present around bearing circumference. More complex FE modelling was done to estimate the deformation of the foils rather than assuming it like an elastic foundation. And the observation predicted that the coefficients against excitation frequency indicated that FE modelled coefficients are less than the coefficients obtained from simple elastic foundation model.

Muruganandam et al. [13] designed gas foil bearing suitable for a high speed micro gas turbine. Mathematical modelling is carried out to apply symmetrical loading to turbine blades and to investigate dynamic motion numerical simulation has been deployed. The rotor is supported by a set of GFBs which are having a cylindrical bush and of radial configuration. For the shaft system Reynold's equation is solved to get air pressure profile in the fluid film. Load capacity is derived by integrating pressure profile over the bearing surface. Displacement velocity and dynamic components are used to calculate damping and stiffness coefficient of the gas foil bearing.

San Andres L [14] investigated Hydrodynamic Fluid Film Bearings and Their Effect on the Stability of Pivoting Machinery with the static and dynamic execution attributes of short length cylindrical journal bearing, with application to the dynamic constrained execution of a unbending rotor upheld on plain orientation. In a radial bearing, the Sommerfield number characterizes a relationship between the static force and the bearing eccentricity inside the bearing. These liquid film direction additionally present goeey damping that helps in diminishing the adequacy of vibrations in working hardware. Reynolds mathematical statement for bearing was inferred which portrays the sum of hydrodynamic weight inside the bearing.

Schiffmann et al. [15] investigated Foil Bearing Design Guidelines for Improved Stability. A lower order foil bearing model, coupled with an inflexible body, linear rotor element model, was utilized to research the basic rotor dynamic instruments and the onset speed of insecurity of a foil bearing-upheld by the rotor.. A sensitivity analysis exhibits that auxiliary damping does not altogether change the onset of sub-synchronous whirl. It is demonstrated, on the other hand, that the introduction of the pivotal food line of the top foil can firmly impact the bearing load limit and rotor dynamic execution.

2.2 MOTIVATION AND OBJECTIVE

Based on the literature review it is quite clear that many works have been carried out and completed in analysis of gas foil bearings. For the fluid film thickness (in this case air) the help of Reynold's equation is used and subsequently using finite difference method or similar numerical methods the pressure profile is calculated. And using this pressure values dynamic coefficients of a foil bearings like stiffness and damping coefficients are calculated. In most of the models developed as of now top foil and bump foil is assumed as an elastic foundation structure and its static stiffness is often ignored.

Here an attempt is made to design CAD model of a bearing and analyze it to obtain its static deflection under the influence of a normal load. After obtaining deflections at multiple loads one can easily obtain static stiffness.

MAIN OBJECTIVES ARE:

1. Develop CAD models of a bump-type gas foil bearing.
2. Finite element analysis of GFBs.
3. Calculate stiffness values for different GFBs

CHAPTER 3: MODELLING AND METHODS

3.1 BUMP-TYPE FOIL BEARING DESCRIPTION

Data for 3 bump type foil bearings are obtained and are mentioned in fig 3. The main components are:

- i. Bearing sleeve: Top foil and bump foil are being supported by a rigid cylindrical ring called bearing sleeve.
- ii. Top foil: Thin flat metal sheet connected to the bearing sleeve. It is free at one end and attached to the sleeve at the other.
- iii. Bump foil: A configuration comprising multiple strips each with multiple bumps. Like top foil one end is attached to the sleeve whereas the other end is kept free.
- iv. Weld: at the fixed end both the foils are attached to the sleeve by means of spot weld to restrict movement of both foils at fixed end.
- v. This sort of bearing allows shaft rotation in only one direction. Initially back of the top foil is in contact with the bump foil and other side with the shaft. As the lift occurs due to the rotor rotation top foil expands and air film is created between the shaft and the foil.

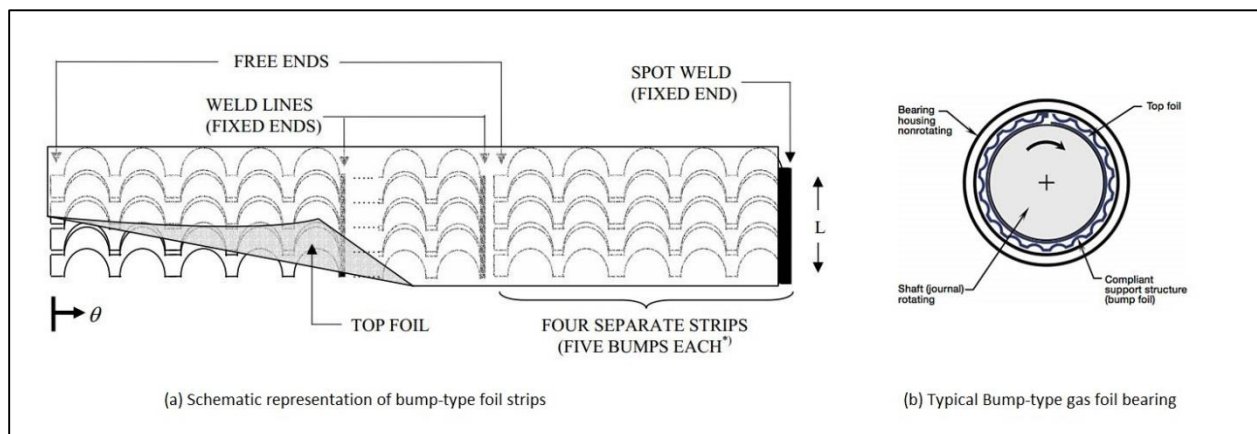


Figure 3: Schematic view of extended bump strips and a typical bump foil bearing [16].

PARAMETERS AND DIMENSIONS:

Table1: Parameters and dimensions of 3 gas foil bearings to be modelled and analyzed

Serial no.	Parameter	GFB1	GFB2	GFB3
1	Journal diameter	28.5mm	25mm	30mm
2	Journal length	28.5mm	25mm	30mm
3	Film clearance	.020mm	.020mm	.020mm
4	Foil thickness (top & bump)	127 μ m	150 μ m	300 μ m
5	Number of bumps	40	36	40
6	Bump height	580 μ m	580 μ m	580 μ m
7	Bump pitch	1.2mm	1.3mm	1.15mm
8	Bump diameter	1mm	1mm	1mm
9	Bump foil Young's modulus	200Gpa	200Gpa	200Gpa
10	Bump foil Poisson's ratio	0.31	0.31	0.31

3.2 DEVELOPMENT OF CAD MODELS

The procedure for creating 3D model of GFB 1 is being shown here. Among all the components the most critical one is the bump structure which is drawn in a 3D modelling software. First of all few curves are drawn and extruded up to the bearing length and then a circular array of the solid generated is created around Z-axis to obtain the complete structure. While drawing the bump foil the dimensions such as pitch height and diameter are referred from Table 1.

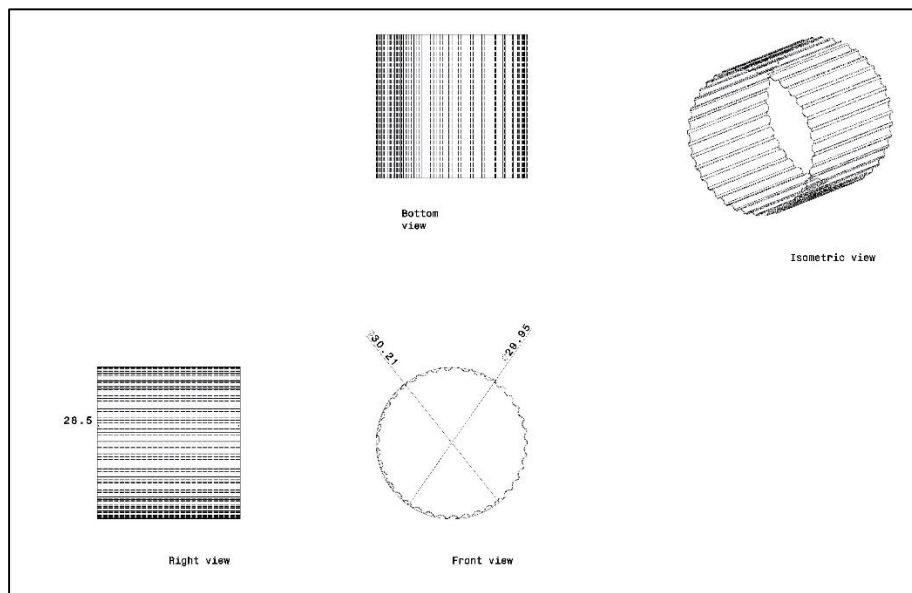


Figure 4: Bump foil structure (all dimensions are in mm).

The next component to draw is the top foil. First of all a circle of desired dimensions drawn and slight trimming done on upper side of the foil to show the gap and one of the open side is extended to connect with the bearing sleeve. Then the sketch is extruded to get 3D model of the top foil.

The next part is the bearing sleeve for which the sketch is just a circle with desired dimensions from Table 1. Once the sketch is drawn a thick cylinder is generated by extruding the sketch by providing a thickness of 2mm.

Similarly the shaft diameter is provided to a circle and extruded to the same length to generate the shaft in CAD environment.

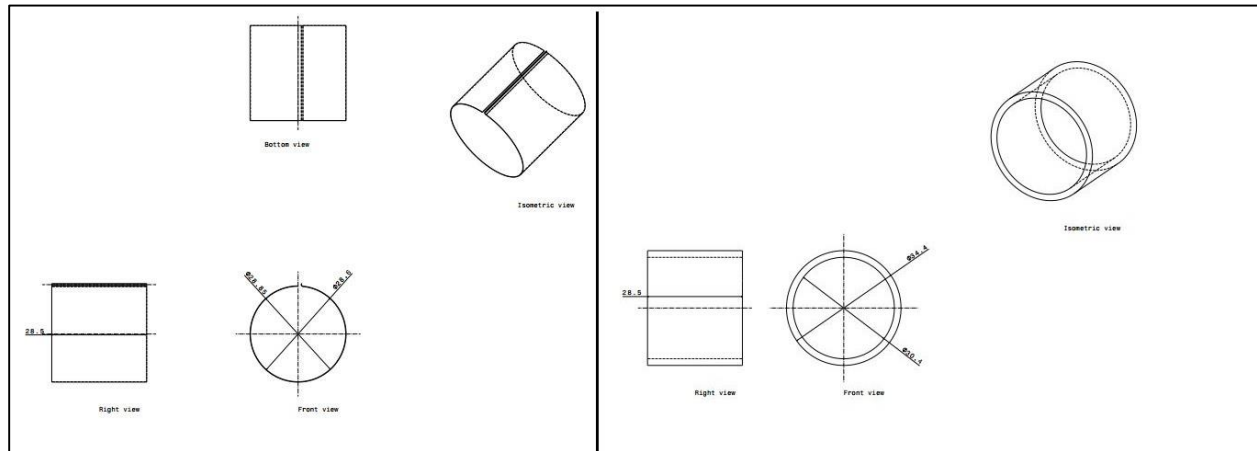


Figure 5: (a) Top Foil structure and (b) Bearing Sleeve (all dimensions are in mm).

Once all the part are generated all are assembled keeping the central axis of the structure in mind and the total axial length. The complete assembly of the bump-type GFB is shown here with exploded view.

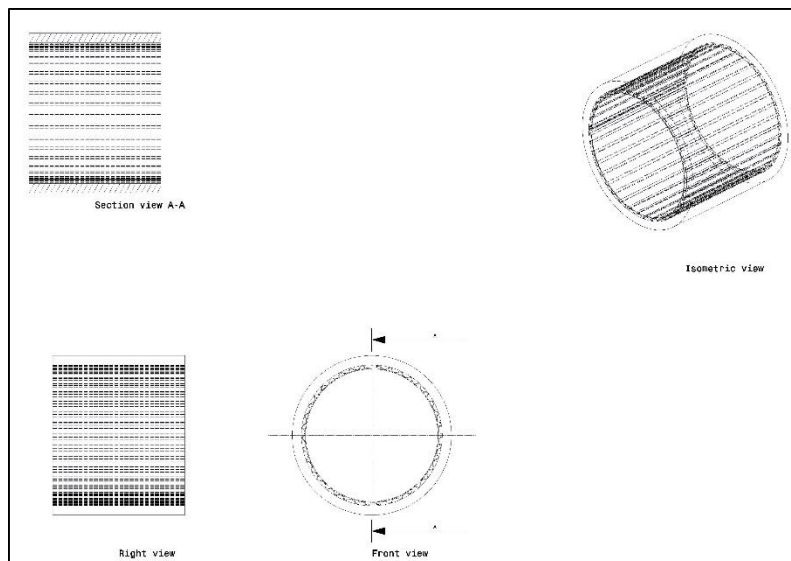


Figure 6: Complete bump-type gas foil bearing (all dimensions are in mm).

3.3 GENERATION OF MESH

Generating mesh is the most important and critical work in engineering simulation in Ansys software. Large no of cells may take longer time to produce solutions without increasing accuracy whereas very few number of cells might produce inaccurate results. Ansys meshing technology provides automated way of generating adequate no of mesh to provide optimum solution. Once the CAD model of the GFB is completed it is imported to the Ansys environment for further simulation. To obtain adequate no of cells automatic mesh is generated. Further relevance center is changed and tetrahedron elements percentage increased to a significant number in order to achieve smooth surface and better mesh quality. The changes in mesh quality is shown below.

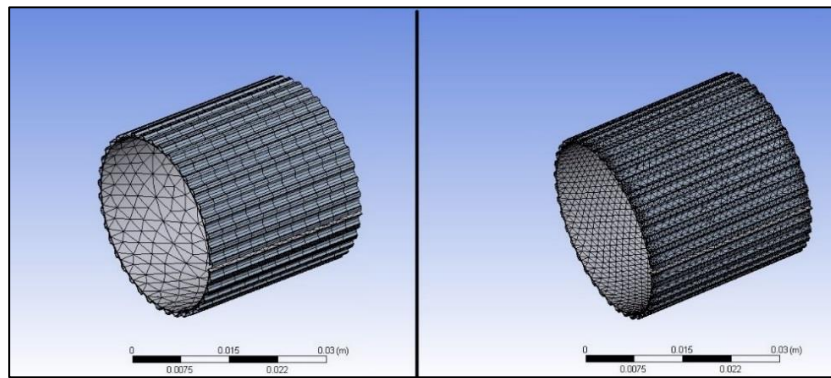


Figure 7: Optimization of auto-generated mesh.

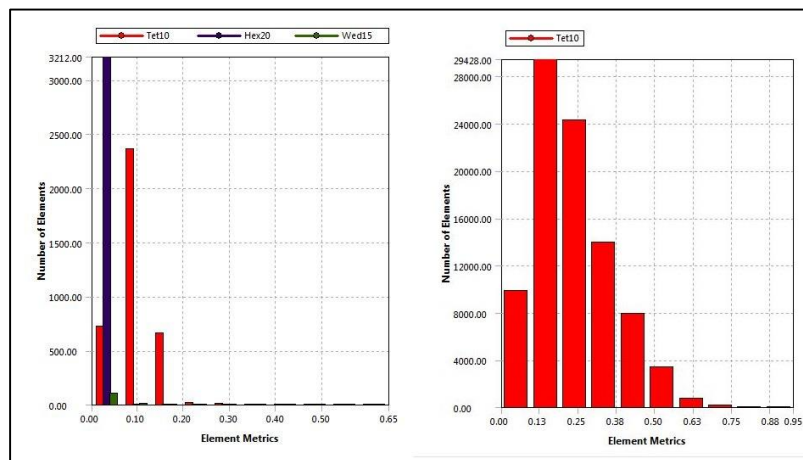


Figure 8: Improvisation of element quality generated by automatic mesh.

3.4 METHOD FOR STIFFNESS CALCULATION

After generation mesh various supports and loads are to be given to these GFBs for calculations of respective deflections.

3.4.1 CALCULATION OF APPLIED LOAD:

The working pressure of a bearing is estimated which lies around 0.1N/mm^2 to 1N/mm^2 . Now to estimate the bearing load the following formula is used:

$$F_1 = P_a \times \pi \times L \times D / 2.$$

Where

F_1 = Bearing load to be applied on top foil in N.

P_a = Working pressure of a bearing in N/mm^2

L = Length of the journal in m.

D = Journal Diameter in m.

The pressure is taken in 10 steps (0.1, 0.2...1.0) and subsequent load values for forces were calculated for all three bearings.

Applied loads:

Table 2: Bearing loads for different GFBs for different pressure values.

Pressure	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
F1(GFB1)	128	255	383	510	638	766	893	1020	1150	1280
F2(GFB2)	98.2	196.4	294.5	392.7	490.9	589.1	687.2	785.4	883.6	981.75
F3(GFB3)	141	283	424	565	707	848	990	1130	1270	1410

3.4.2 APPLYING BOUNDARY CONDITIONS AND LOAD:

Bearing load is applied to the top foil according to table 2 and the load is applied along the X-axis. The bearing sleeve is given a cylindrical support as a constraint in the model. The modelling is done in static structural and the time step is given as 1 sec. solver controlled are dealt by program controllers and large deflection and inertia relief set to zero.

Bonded connections are given between following pairs:

- i. Bearing sleeve and bump foil
- ii. Bearing sleeve and top foil (at the fixed end of the top foil)
- iii. Top foil and bump foil

Following results are to be obtained from the static structural analysis of above mentioned 3 gas foil bearings:

- i. Total deformation
- ii. Directional deformation along X-axis
- iii. Directional deformation along Y-axis
- iv. Equivalent stress (Von-Mises stress)

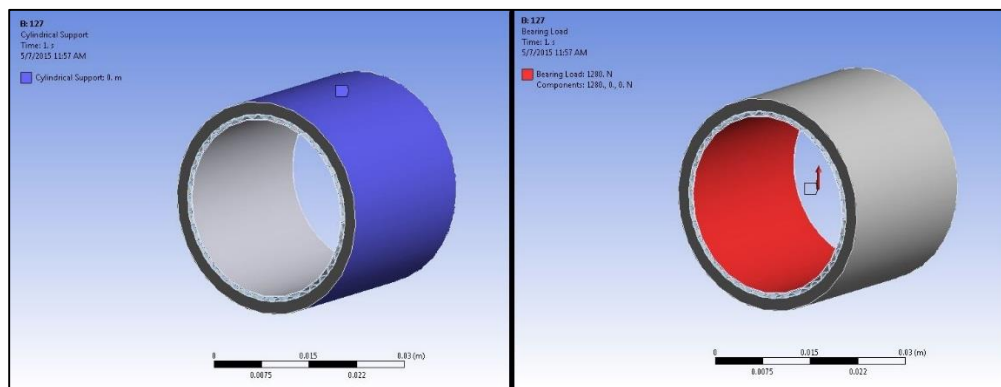


Figure 9: Applying boundary conditions and bearing load to GFB1 for pressure corresponding to 1N/mm^2 .

3.5 METHOD FOR TEMPERATURE DEPENDENT STIFFNESS CALCULATION

As the shaft rotates within the foil bearing the temperature of fluid film increases due to friction. As a result the temperature of the top and bottom foil increases due to convection. Increased temperature of the foils changes the stiffness of the foil bearing hence it is necessary to investigate effect of temperature over stiffness.

To calculate effect of temperature over stiffness a bearing load corresponding to 1N/mm^2 is applied and the temperature is varied from 10^0C to 100^0C . Cylindrical support is provided to the bearing sleeve and bearing load is applied to the top foil. The thermal conditions are applied to the top foil and bump foil as well. The ambient temperature is set at 22^0C .

Set up for GFB 1 at 1280 N load and 10^0C is shown here:

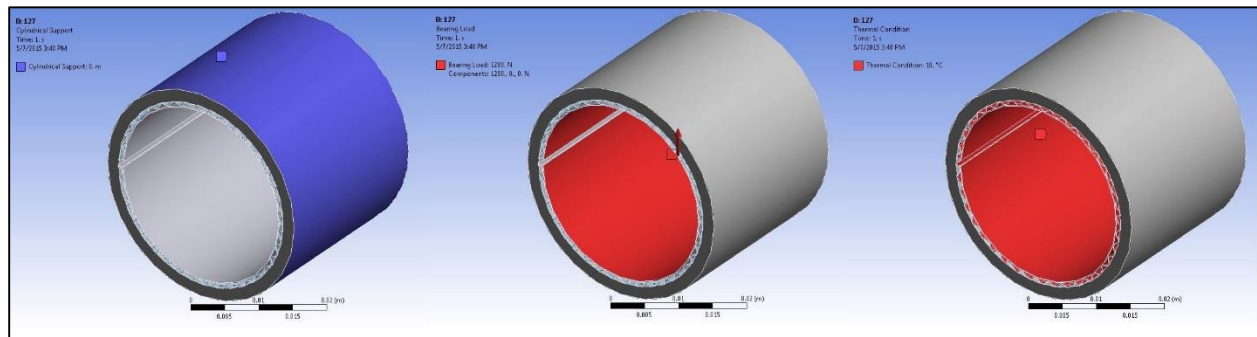


Figure 10: Applying boundary conditions, thermal conditions and bearing load to GFB1 for pressure corresponding to 1N/mm^2 and temperature 10^0C .

CHAPTER 4: RESULT AND DISCUSSION

4.1 CALCULATION OF STIFFNESS

Ansysis simulation of all 3 gasfoil bearings for a varying pressure range (0.1-1 N/mm²) is carried out and total deformation, directional deformation and von-Mises stress are obtained. Results for GFB 1 at 1280N are shown here.

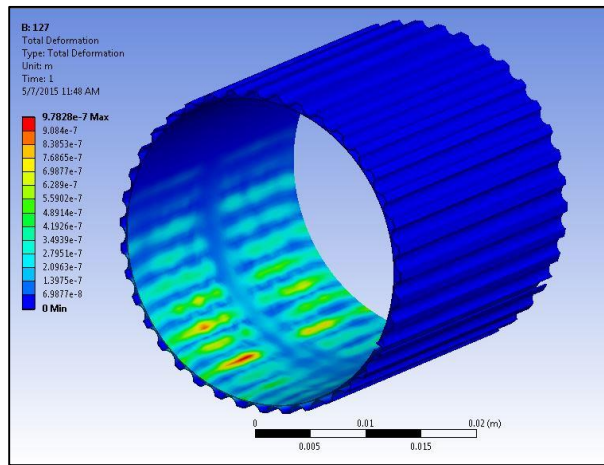


Figure 11: Variation of total deformation.

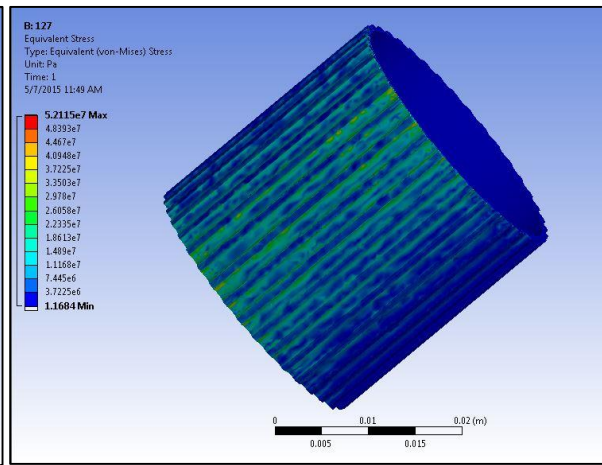


Figure 12: Variation of equivalent stress.

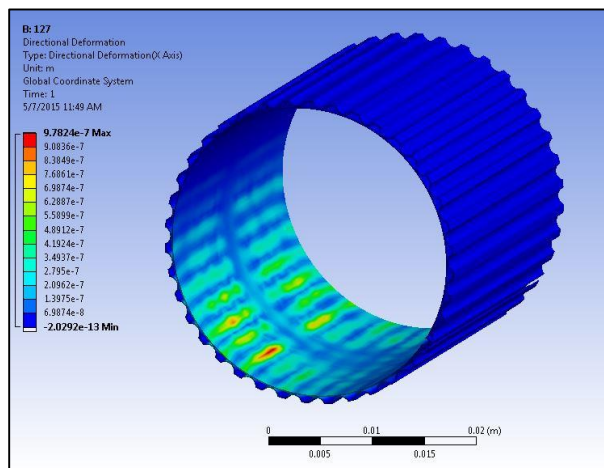


Figure 13: Directional deformation (X-axis).

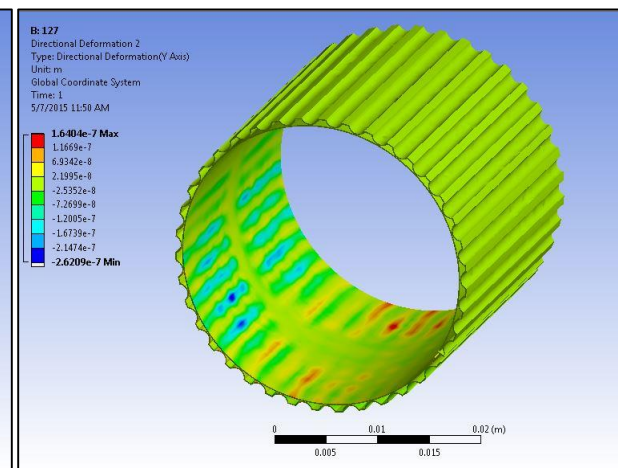


Figure 14: Directional deformation (Y-axis).

Table 3: Bearing load and directional deformation (along X-axis) for a varying pressure range.

Serial no.	F1(GFB1) in N	F2(GFB2) in N	F3(GFB3) in N	X1(GFB1) in mm	X2(GFB2) in mm	X3(GFB3) in mm
1	128	98.17	141	4.36×10^{-8}	2.31×10^{-8}	1.04×10^{-8}
2	255	196.35	283	8.69×10^{-8}	4.61×10^{-8}	2.09×10^{-8}
3	383	294.52	424	1.31×10^{-7}	6.92×10^{-8}	3.12×10^{-8}
4	510	392.70	565	1.74×10^{-7}	9.22×10^{-8}	4.16×10^{-8}
5	638	490.87	707	2.18×10^{-7}	1.15×10^{-7}	5.21×10^{-8}
6	766	589.05	848	2.61×10^{-7}	1.38×10^{-7}	6.25×10^{-8}
7	893	687.22	990	3.04×10^{-7}	1.61×10^{-7}	7.30×10^{-8}
8	1020	785.40	1130	3.48×10^{-7}	1.84×10^{-7}	8.33×10^{-8}
9	1150	883.57	1270	3.92×10^{-7}	2.08×10^{-7}	9.36×10^{-8}
10	1280	981.75	1410	9.78×10^{-7}	2.31×10^{-7}	1.04×10^{-7}

PLOTS FOR BEARING LOAD VS DEFLECTION:

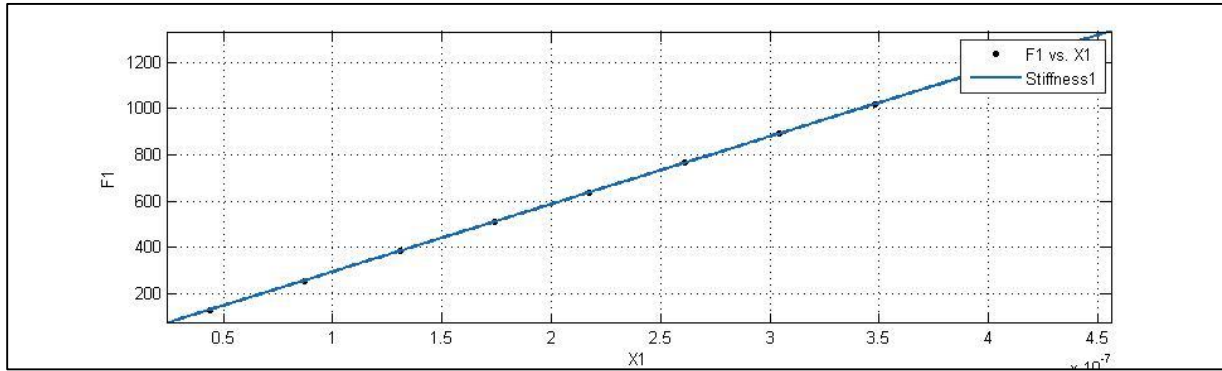


Figure 15: Bearing load (N) vs Deflection (mm) for GFB 1, (F1 vs X1).

Equation of curve $F1 = K_1 \times X1 + C_1$

Coefficients (with 95% confidence bounds):

$$K_1 = 2.927 \times 10^9 (2.92 \times 10^9, 2.934 \times 10^9) \quad C_1 = 0.8717 (-1.074, 2.817)$$

Stiffness of GFB 1 is found to be **$K_1 = 2.927 \times 10^9$ N/mm.**

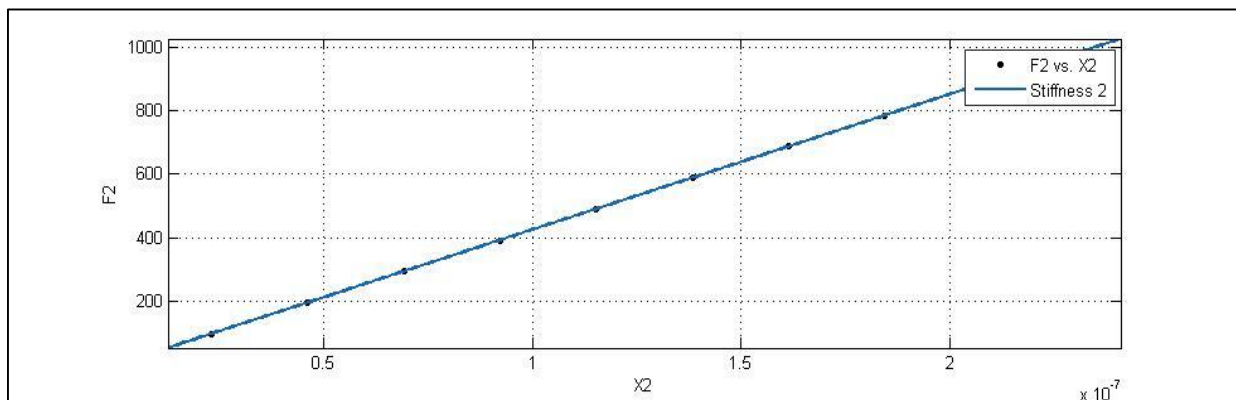


Figure 16: Bearing load (N) vs Deflection (mm) for GFB 2, (F2 vs X2)

Equation of curve $F2 = K_2 \times X2 + C_2$

Coefficients (with 95% confidence bounds):

$$K_2 = 4.258 \times 10^9 (4.257 \times 10^9, 4.258 \times 10^9)$$

$$C_2 = 0.003786 (-0.003053, 0.01063)$$

Stiffness of GFB 2 is found to be **$K_2 = 4.258 \times 10^9 \text{ N/mm}$** .

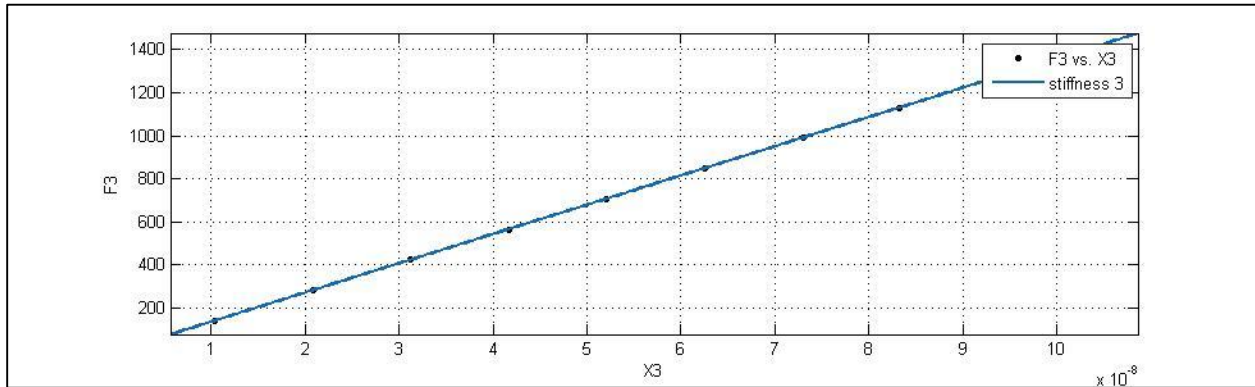


Figure 17: Bearing load (N) vs Deflection (mm) for GFB 3, (F3 vs X3)

Equation of curve $F3 = K_3 \times X3 + C_3$

Coefficients (with 95% confidence bounds):

$$K_1 = 1.36 \times 10^{10} (1.358 \times 10^{10}, 1.362 \times 10^{10})$$

$$C_1 = -0.9329 (-2.493, 0.6273)$$

Stiffness of GFB 3 is found to be **$K_3 = 1.36 \times 10^{10} \text{ N/mm}$** .

4.2 VARIATION OF STIFFNESS WITH TEMPERATURE

Ansys simulation of all 3 gasfoil bearings for a fixed pressure 1N/mm^2 is carried out and total deformation, directional deformation and von-Mises stress are obtained. Results for GFB 1 at 1280N are shown here.

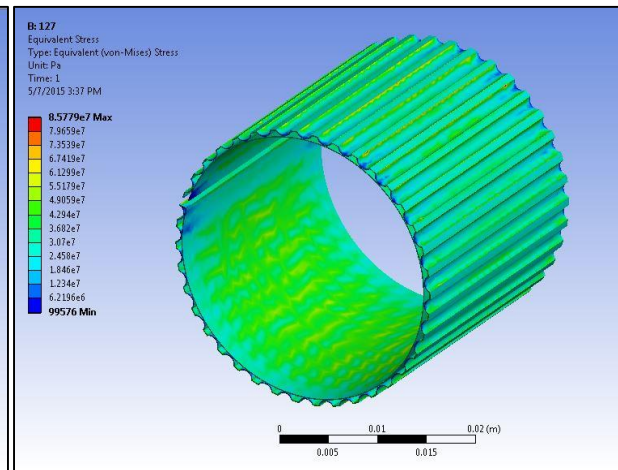
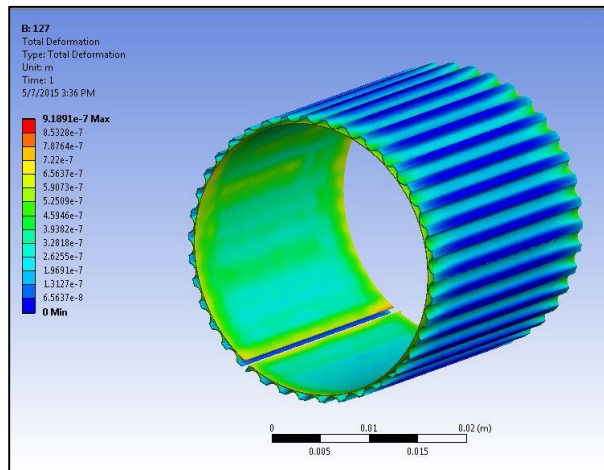


Figure 18: Variation of total deformation at 10°C. Figure 19: Variation of equivalent stress at 10°C.

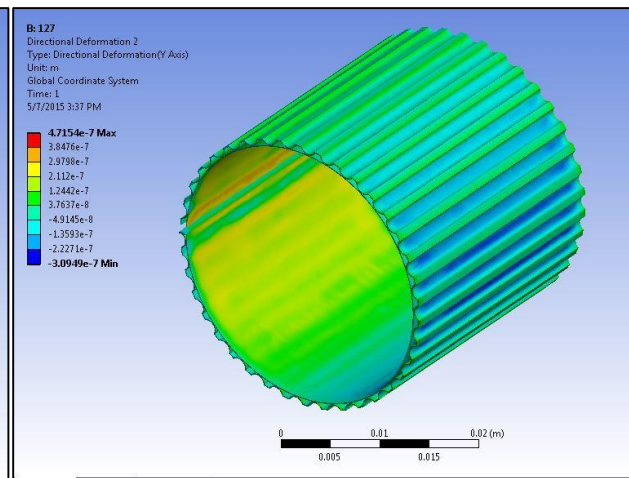
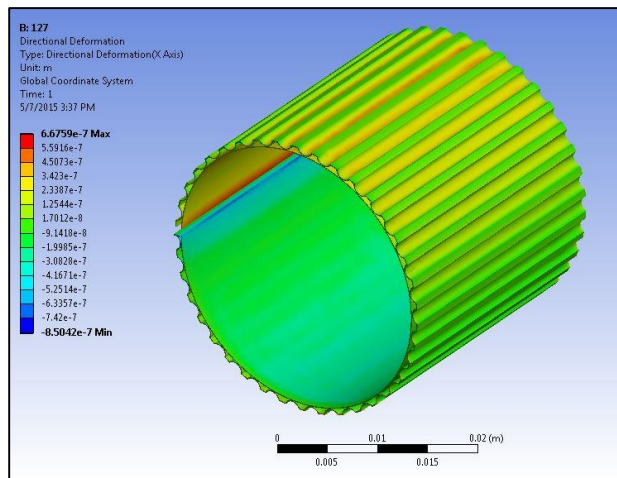


Figure 20: Directional deformation (X-axis) at 10°C. Figure 21: Directional deformation (Y-axis) at 10°C.

Table 4: Temperature and directional deformation (along X-axis).

Serial no.	Temperature In $^{\circ}\text{C}$	Deformation in GFB1 at 1280N (in mm)	Deformation in GFB2 at 981.75 N (in mm)	Deformation in GFB3 at 1410N (in mm)
1	10	6.68×10^{-7}	8.88×10^{-7}	7.21×10^{-7}
2	20	4.25×10^{-7}	4.63×10^{-7}	1.51×10^{-7}
3	30	5.76×10^{-7}	2.31×10^{-7}	6.59×10^{-7}
4	40	1.29×10^{-6}	6.39×10^{-7}	1.48×10^{-6}
5	50	2.00×10^{-6}	1.28×10^{-6}	2.30×10^{-6}
6	60	2.72×10^{-6}	1.92×10^{-6}	3.12×10^{-6}
7	70	3.43×10^{-6}	2.55×10^{-6}	3.95×10^{-6}
8	80	4.14×10^{-6}	3.19×10^{-6}	4.77×10^{-6}
9	90	4.86×10^{-6}	3.83×10^{-6}	5.59×10^{-6}
10	100	5.57×10^{-6}	4.47×10^{-6}	6.41×10^{-6}

PLOTS FOR TEMPERATURE VS STIFFNESS:

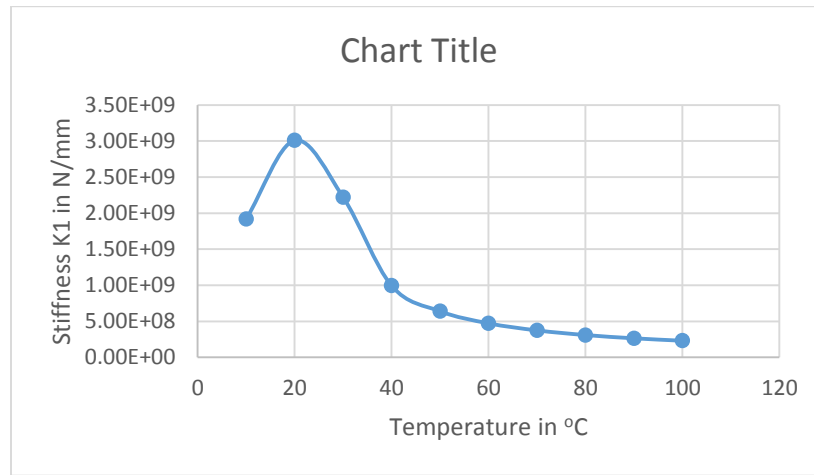


Fig 22: Stiffness (N/mm) vs Temperature (°C) for GFB 1.

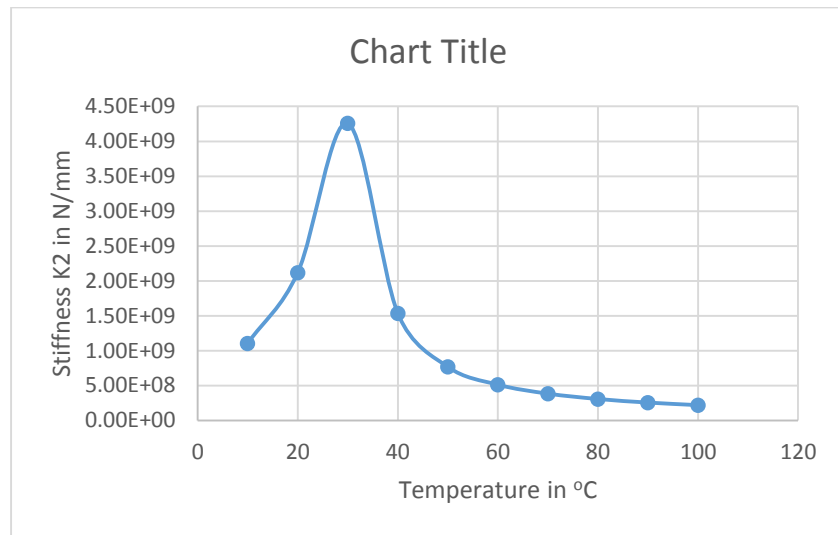


Fig 23: Stiffness (N/mm) vs Temperature (°C) for GFB 2.

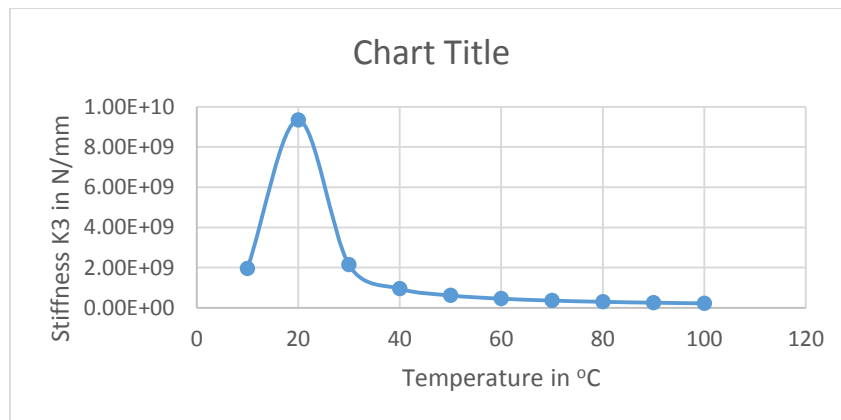


Fig 24: Stiffness (N/mm) vs Temperature (°C) for GFB 3.

CONCLUSION

Bearing load is applied to 3 different models of gas foil bearings. Deflections for different load values obtained using finite element analysis. Graph is plotted between load and deflection and a straight line is obtained, the slope of each curve represents stiffness of each bearing.

Graph between stiffness and temperature is plotted and the variation is obtained. It is observed that the stiffness is maximum as the temperature of the bearing coincides with the ambient temperature and stiffness reduces as the temperature difference between the bearing and working environment increases.

FUTURE SCOPES:

In this report stiffness due to static load and its variation according to working temperature is obtained. Based on the results obtained following works can be done in future:

- i. Pressure variation due to hydrodynamic effect of fluid film can be calculated and incorporated to the designed model to obtain dynamic coefficient and both these stiffness values will provide the overall stiffness of the gas foil bearing.
- ii. Similarly damping coefficients can be calculated for any air foil bearing.
- iii. Once both the dynamic coefficients (stiffness and damping) are obtained those values can be fed to rotor-dynamics simulation for vibration analysis and determination of critical speed.

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